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The Concept of New-Generation Steam Turbines for Coal Power Engineering of Russia. Part 1. Economic and Technical Substantiation of the Concept

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Abstract—Development of the concept of designing modern steam turbines and its application to turbines for ultrasupercritical steam conditions are considered. The results from predraft designing of a turbine for ultrasupercritical steam conditions with a capacity of around 700 MW in a two-cylinder version that corresponds to this concept are presented. Main problems relating to construction of such turbines under the conditions of Russia are analyzed.

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Heat-recovery type combined-cycle plants (HR-CCPs) (in case of using gaseous fuel) and traditional steam-turbine units (STUs) (in case of using solid fuel) will remain the best heat engines for modern thermal power stations (TPSs) in the nearest and more distant future.

Not only are modern HR-CCPs the most efficient prime movers for large-scale power engineering (their electrical efficiency is at present as high as 60%), but they are also characterized by the lowest specific capital outlays.

The reserves of natural gaseous and liquid fuel are limited; in any case, their amounts are hundreds if not thousands of times smaller than those of solid fuel. Therefore, transition for using predominantly solid fuel (and nuclear energy) in power engineering is historically unavoidable. Combined-cycle plants firing solid fuel in a pressurized furnace (CCP-S) and with gasification of solid fuel (CCP-G) have not yet become commercially available in the power engineering market and are in the stage of perfecting experimental-industrial models [1, 2]. These installations will become competitive with traditional STUs in a more distant future (in all likelihood, in 15–20 years or more).

The current period of time is characterized by the fact that traditional steam-turbine power units operating on solid fuel are intensely developed in leading countries. These power units are being improved in the directions of achieving higher parameters of steam, improving the thermal circuit, decreasing all sorts of heat losses in boilers, turbines, generators, and their equipment, and working out optimal design solutions.

Below, the present state of STUs for ultrasupercritical steam conditions is considered together with the currently available possibilities of constructing such an STU that would have perfect efficiency and design. The typical indicators of such units presented, e.g., in [1] are as follows:

Turbine capacity, MW	575–700
Initial pressure, MPa	30
Steam temperature, °C	
downstream of the boiler	600
downstream of the steam reheater	600
Feedwater temperature, °C	302
Efficiencies of	
turbine unit (net)	0.49
boiler	0.93
power unit	0.46
Heat rate at a net efficiency of 0.49, kJ/(kW h)	7347

The main characteristics of existing power units for ultrasupercritical steam conditions and those being designed are presented in [2]. Power units operating at parameters close to those given above have approximately the same efficiency. For example, the 740-MW power unit installed in 1998 at the Hessler TPS in Germany, which operates at the parameters $t_0/t_{s.r}$ = 580/600°C, p_0 = 27.5 MPa, t_{fw} = 301°C, and p_c = 3.6 kPa, has a net efficiency equal to 0.454, and the Alborg power unit (commissioned in 1997 in Denmark), which operates at $t_0/t_{s.r1}/t_{s.r2}$ = 580/580/600°C, p_0 = 28.5 MPa, t_{fw} = 300°C and high vacuum (p_c = 2.35 kPa), has efficiency equal to 0.49. In [3], Siemens gives close values: its power unit, operating at p_0 = 30 MPa, $t_0/t_{s.r1}/t_{s.r2}$ = 600/600/600°C, and p_c = 4 kPa, has a net efficiency equal to 0.45, where t_0, $t_{s.r1}$, and $t_{s.r2}$ are the temperatures of live steam and the first and second steam

reheating, p_0 is the live steam pressure, p_c is the pressure in the condenser, and t_{fw} is the feedwater temperature.

The list of main parameters that determine the power unit efficiency includes p_0, t_0, $t_{s.r}$, and p_c.

Modern power units for ultrasupercritical steam conditions are designed for the maximal live steam and reheat steam temperature equal to 600–620°C, which is determined by using medium-alloy carbon steels with the content of main alloying element (chromium) equal to 9–12% and with addition of many other alloying elements (molybdenum, niobium, tungsten, vanadium, boron, cobalt, and others) in small quantities as material for making the high-temperature elements of boilers and turbines.

At present, programs exist in Europe and the United States for developing next-generation coal-fired power units for the pressure p_0 = 38 MPa and temperatures $t_0/t_{s.r1}/t_{s.r2}$ = 700/720/720°C, the construction of which will generate the need to use other, more heat-resistant materials for the elements of boilers and turbines. It is expected that the efficiency of such a power unit will be higher than 0.50. Still higher values of t_0 and $t_{s.r}$ can be achieved only through the use of more heat-resistant materials. If we assume the possibility of using nickel alloys (nimonics) for the high-temperature elements of boilers and turbines (for boilers this is a must, while in turbines internal steam cooling can in principle be organized), the maximum steam temperature in power units for ultrasupercritical steam parameters will reach 800°C.

An assessment of the efficiency of a turbine unit with the parameters p_0 = 38 MPa, $t_0/t_{s.r1}$ = 800/800°C, and p = 3.5 kPa gave the gross efficiency $\eta_{TU}^{gr} \approx 0.57$. If we take the consumption of energy for the power unit auxiliaries equal to 5% (considering high pressure of the feedwater pump) and the boiler efficiency η_b = 0.94, the power unit efficiency will be $\eta_{p.u} \approx 0.51$. It should be pointed out that the steam quality x_b in the considered example is equal to 0.96. The use of the second steam reheating at the temperatures $t_0/t_{s.r1}/t_{s.r2}$ = 800/800/800°C results in that the turbine unit efficiency decreases to approximately 0.56 because in this case superheated steam with the temperature $t_c \approx$ 153°C will enter into the condenser.

European producers (Alstom, Siemens, Babcock, Borsig, and others) have been working on the AD700 project since 1994 [4]. The AD700 power unit has the following parameters: p_0 = 35 MPa, $t_0/t_{s.r}$ = 700/720°C, t_{fw} = 350°C, p_c = 4.0 kPa, the number of extractions for regeneration is equal to 8, and the power unit capacity is equal to 600 MW. The expected efficiency of the power unit is equal to 0.50–0.51 during its operation with cooling towers, and 0.53–0.54 in the case of its operation with two steam reheating stages (with cooling by seawater). It is planned that the pilot power unit will be commissioned in 2011. The program under which the AD700 project is developed takes financial support from the participants themselves, whereas money for programs developed in Japan and the United States are paid fully from the state budget.

These data are only tentative; nonetheless, they show that a noticeable improvement in the efficiency of a steam power unit can hardly be expected even for steam temperatures of 800/800/800°C, whereas the cost of such a power unit will be considerably higher.

Traditional steam power units will never overtake modern and the more so advanced HR-CCPs in efficiency. Therefore, efficient use of solid fuel is possible only in SSP-Ss and CCP-Gs, which are not available in the market of power equipment yet.

THE CONCEPT OF A TURBINE FOR ULTRASUPERCRITICAL STEAM CONDITIONS

Efficiency and cost of 1 kW of installed capacity are the most important characteristics of the considered generation of power units for ultrasupercritical steam conditions. The turbine itself accounts for a comparatively small fraction of specific cost, which seems to be at a level of 15–20%. Nonetheless, efforts should be taken to reduce the cost of the turbine, and selection of the optimal turbine design is the main factor in solving this problem. Apart from featuring lower cost of manufacturing, the optimal turbine design should meet such requirements as high reliability, long period of time between overhauls, good repairability (characterized by low costs of repairs), and acceptable maneuverability.

An analysis and generalization of experience gained in Russia and abroad in design, construction, and operation of power units for supercritical pressure and ultrasupercritical stream conditions makes it possible to formulate the main principles (concept) for 500–700-MW power units:

—The turbine is made with a combined high- and intermediate-pressure cylinder (HIPC), the rotor of which is cooled in the zone of the intermediate-pressure part's (IPP) first stage.

—All rotors of the turbine, including the low-pressure rotor (LPR), are all-forged ones.

—The rotors of the low-pressure cylinders (LPCs) must be made with external supports (these supports should not be built into the exhaust hoods).

—The turbine set's shaft system must be made with one support between the adjacent rotors.

—The system of thermal expansions must be designed so that the shaft system supports are rigidly

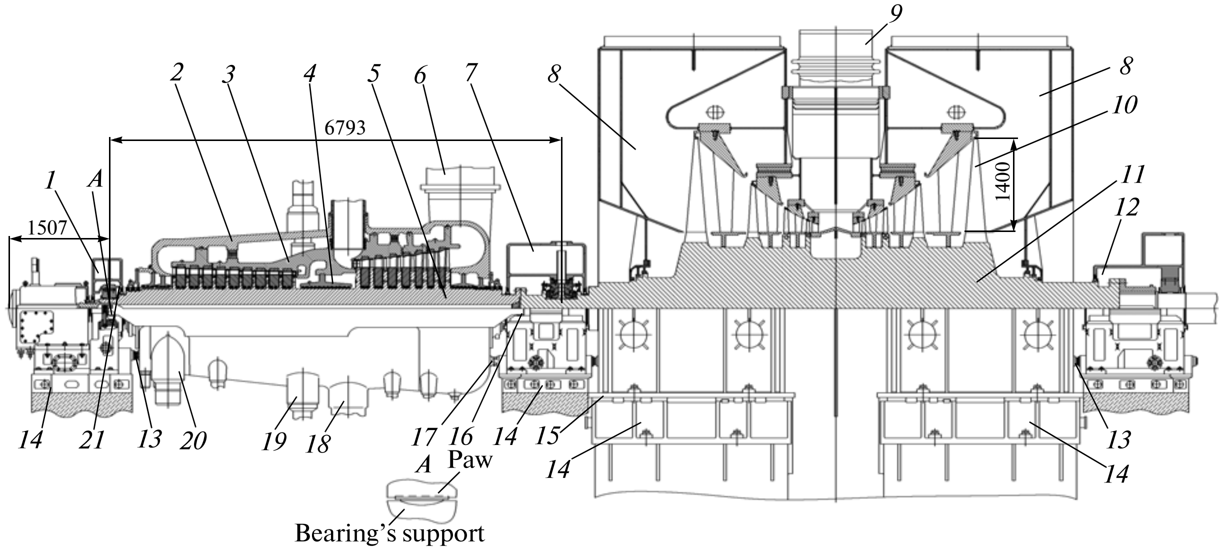

Fig. 1. Longitudinal section of the K-660-30 turbine for ultrasupercritical steam conditions. (*1*), (*7*), and (*12*) casings of the first, second, and third bearings; (*2*) and (*3*) external and internal casings of the HIPC; (*4*) middle seal of the HIPC; (*5*) and (*11*) rotors of the HIPC and LPC; (*6*), (*8*), and (*20*) exhaust hoods of the IPP, LPC, and HPP; (*9*), (*18*) and (*19*) steam admission sockets of the LPC, IPP, and HPP; (*10*) last-stage rotor blades; (*13*) vertical keys; (*14*) foundation frames; (*15*) LPC fixing point; (*16*) and (*21*) rear and front support paws of the HIPC; (*17*) loop joint of the HIPC and second bearing's shell (the HIPC fixing point). Detail *A* shows the resting of the HIPC paws on the bearing pedestal.

secured on the foundation frames and the cylinder paws can freely (without keys) slide over the support bodies.

—The turbine should be furnished with one double-flow LPC, the last-stage buckets of which should be made of titanium allow and have an adequate flow area.

The schematic drawing of a turbine meeting the above-mentioned concept is shown in Fig. 1.

Of course, the aforesaid principles for construction of turbine units are not limited to turbine units for ultrasupercritical steam conditions and can (should) be applied to installations of smaller capacity and for standard (subcritical and supercritical) steam conditions [5].

Below, the advantages of turbine units meeting this concept are considered.

The use of a combined HIPC allows the turbine design to be made much simpler: it has a fewer number of bearings, shells, and end seals; the turbine has a shorter length; and it becomes easier to organize the freedom of turbine thermal expansions on its foundation. On the whole, the use of a combined HIPC makes it possible to reduce the cost of the turbine and achieve essential improvement in its operational reliability. A combined HIPC can be constructed with almost no loss to its efficiency as compared with the version made with separate high- and intermediate-pressure cylinders (HPC and IPC). Many advanced manufacturers use a combined HIPC; in particular, GE Energy uses such a solution in its turbines for supercitical steam conditions with capacities of up to 650 MW.

The choice of the type of a low-pressure rotor (LPR) is mainly determined by the technological traditions of the manufacturer and plays an important role in shaping the LPC flow path, because it determines the possibility of increasing the root diameter. Assembled LPRs operate with increased stresses in their disks and are generally more susceptible to corrosion cracking than other types of rotors. All-forged LPRs made without a central hole have high strength and resistance to corrosion, but they have technological limitations on their external diameter. Welded rotors seem to be the most suitable choice for LPCs. Below, an all-forged LPR is considered as an article the production of which has been mastered best by Russian manufacturers.

It is more preferable to use external bearings instead of built-in ones, because they have stiffer supports and make it possible to implement the principle of using one support between the adjacent rotors. With this principle implemented, it becomes possible to eliminate almost completely the occurrence of dangerous misalignments, which lead to serious consequences if two bearing bushings are installed close to each other in one support. With the occurrence of misalignments eliminated, better conditions are created for keeping the reactions in supports at constant levels, for normal operation of the bearings, and for keeping the shaft system's vibration characteristics (critical

rotation frequencies, threshold power, and threshold rotation frequency) unchanged.

Naturally, the design in which one support is used between the adjacent rotors allows the turbine unit to be made with a fewer total number of bearings, which in itself helps improve its operational reliability. In addition, with such a design the shaft system has a fewer number of natural frequencies lying in the band of working rotation frequencies, due to which it becomes easier to detune the shaft system from resonances.

The improved system of thermal expansions (see Fig. 1) has the following specific features.

First, all supports of the shaft system are rigidly fixed on the foundation frames (without sliding).

Second, the HIPC casing rests freely on the protrusions made in the pedestals of the first and second bearings using the front and rear paws through polished wear-resistant highly rigid pads (leader *A* in Fig. 1). In this case, the paws can freely (without any keys) and with low friction resistance slide in the longitudinal and transversal directions when the HIPC outer casing experiences thermal expansion. In this design, the friction on the sliding surfaces is considerably lower than it is in the system traditionally used in Russian turbines, in which the sliding surfaces of bearing pedestals contacting with the foundation frames experience the weight of cylinders, rotors, and bearing pedestals. In the new design, only the weight of the HIPC casing is transferred to the sliding surfaces, and the surfaces themselves have small sliding areas, due to which it becomes much easier to fit them and freedom of motion is ensured. Low friction means that only insignificant forces are applied to the foundation girders, and that only small twisting and bending strain occurs in them. The friction force $T = kN$, where k is the friction coefficient and N is the normal force acting on the sliding surface. This design makes it possible to obtain the minimal value of k due to the use of pads with highly rigid and polished sliding surfaces. As was already mentioned, the normal force N has also been minimized.

Third, the HIPC fixing point is organized by using a loop joint between the second casing's support and the external HIPC.

Fourth, the thrust bearing (which is combined with the second bearing) is placed close to the HIPC fixing point, which makes it possible to obtain small relative displacements of the HIPC rotor and casing and, hence, to achieve small changes in the axial gaps between the diaphragms and disks in the high-and-intermediate-pressure part.

Apart from featuring essentially lower friction forces that arise when thermal expansions occur in the HIPC and simpler construction of its supports, the proposed system has some additional advantages:

(i) There is no need to process and fit large areas through which the bearing shells should slide over the foundation frames (the need of using large sliding areas is unavoidable in the traditional scheme).

(ii) The occurrence of misalignment and jamming in longitudinal keys, which is possible in the traditional scheme when the bearing shell slides over the foundation frame, is excluded, because the bearing pedestals are rigidly fixed on the foundation in the new construction.

The LPC rests on the foundation frames according to the traditional scheme, i.e., using lateral balconies. The LPC fixing point is located at the spot in which the axis of transverse keys intersects the turbine meridian plane near the HIPC fixing point and the thrust bearing. With such a solution, the LPC casing and rotor expand in one direction and have small relative displacements.

The proposed design solution for resting the turbine on the foundation [6] allows the problem of ensuring normal thermal expansions to be solved radically. With this design, axial displacements of the rotor with respect to the casing that occur when they experience thermal expansions are minimized, due to which small changes are achieved in the axial gaps between the diaphragms and runners both in the HIPC and LPC.

The turbine shaft system consists of a common high- and intermediate-pressure rotor (HIPR) and an LPR and rests on three bearings. The third bearing should be made as a common one for the LPR and the rotor of the electrical generator (EGR). The turbine unit's entire rotor system composed of the HIPR + LPR + EGR rests on four bearings.

MAIN PROBLEMS RELATING TO DEVELOPMENT OF AN OPTIMAL TURBINE FOR ULTRASUPERCRITICAL STEAM CONDITIONS

Development and construction of the low-pressure part's last stage having the maximally possible throughput capacity is the top priority among the main problems that need to be solved in constructing the optimal steam turbine for ultrasupercritical steam conditions in accordance with the considered concept. The following positive features can be obtained if we succeed in increasing steam flowrate through the last stage at a given power output:

(i) The turbine can be made with a fewer number of low-pressure flows (cylinders).

(ii) The turbine unit can be made with a shorter length and smaller weight.

(iii) The shaft system can be made with a fewer number of supports.

(iv) The turbine can be made with a fewer total number of stages.

By realizing these possibilities we can reduce the cost of the turbine itself, the turbine installation (as a result of using a simplified thermal circuit), and the entire power station due to smaller dimensions of the turbine building and, in particular, the turbine unit foundation. In addition, better operational reliability of the turbine is achieved through decreasing the number of similar elements in the turbine and in its thermal circuit, and due to using a simpler design of the thermal expansion system. Easier construction, a shorter period of time required for carrying out the construction, and lower cost required for erection and repairs of the turbine set are other obvious advantages.

By using a properly designed turbine last stage with an increased throughput capacity, it is possible to achieve higher efficiency of the stage itself and of the entire low-pressure part. The state of developments of last-stage blades with large exhaust area is reflected in Table 1.

The data given in Table 1 testify that the optimal design solution for the K-660-30 turbine is to make it with one double-flow LPC (line 5). This solution differs from version 4 in that the rotor blades in it are shorter by 100 mm (and so is the size of guide vanes at the periphery). The economic efficiency was analyzed exactly for this version. Siemens has also announced designing of a similar blade [this seemed to happen a decade later than this work was commenced by LMZ and the Moscow Power Engineering Institute(MEI)].

Development of such stages with the maximum possible throughput capacity takes a long period of time and requires a large volume of investigations (including calculation studies, which must necessarily be supplemented with experimental studies). A full-scale experimental setup must also be constructed for checking and substantiating the main solutions adopted in the design. However, in our opinion, the end justifies the means, and such a task must be set forth and solved.

Development and construction of the HIPC, which is another important problem, is much easier than the first one. It has been shown in the predraft design of the K-660-30 turbine (see below) that a sufficiently reliable high- and intermediate-pressure rotor can be constructed provided that suitable materials and design solutions are selected.

PREDRAFT DESIGN OF THE K-660-30 TURBINE FOR ULTRASUPERCRITICAL STEAM CONDITIONS[1]

Calculation of the thermal circuit and efficiency. Bearing in mind the achieved level of modern power units for ultrasupercritical steam conditions, we took the following parameters of the turbine for making a calculated assessment of its main characteristics: electrical capacity N_{el} = 660 MW, p_0 = 30 MPa, $t_0/t_{s.r}$ = 610/620°C, p_c = 3.5 or 5.0 kPa, and t_{fw} = 300°C.

The thermal circuit of the turbine unit includes four low-pressure heaters (LPHs), a deaerator, and three high-pressure heaters (HPHs). The feedwater pump is driven by an electric motor (Fig. 2).

The efficiencies of compartments were adopted among the main initial data from the results of preliminary logic assessments. The final efficiencies of the compartments and cylinders can be determined only after the turbine flow path design is worked out and its thermal calculation is performed. A preliminary calculation of the thermal circuit was carried out for roughly estimating the turbine unit's gross heat consumption, the flowrates and parameters of steam in the characteristic places of the turbine, and tentatively estimating the power unit efficiency with accuracy determined by the specified power requirements for the power unit auxiliaries and boiler efficiency.

The results obtained from the thermal calculation of the K-660-30 turbine unit are as follows:

Electric power output (at the generator terminals) N_{el}, MW	669.75
Steam flowrate, kg/s	
at the inlet to the high-pressure part (HPP)	472.3
at the inlet to the IPP	382.0
to the condenser	281.5
Parameters of steam before the nozzles of the HPP's first stage	
pressure, MPa	28.5
temperature, °C	605.5
Parameters of steam before the nozzles of the IPP's first stage	
pressure, MPa	4.5
temperature, °C	619.7
Parameters of steam at the outlet from the LPP's last stage	
pressure, kPa	3.5
steam quality	0.907
Gross heat consumption of the turbine unit, kJ/(kW h)	6852.1

[1] V.A. Fadeev, A.E. Zakharov, and E.V. Klimova took active participation in working out the predraft design.

Table 1. Parameters characterizing last-stage rotor blades from different producers (rotation frequency $n = 3000$ rpm)

No	Producer	Ω, m^2	l, mm	d_m/l	d_r, mm	u_{per}, m/s	Material	Note
1	LMZ	8.8	1000	2.8	1800	597	Steel	In operation
2	LMZ	11.3	1200	2.5	1800	660	Titanium alloy	"
3	LMZ–MEI	16.95	1388	2.8	2500	828	"	In development
4	LMZ–MEI	17.9	1500	2.53	2300	833	"	"
5	LMZ–MEI	16.27	1400	2.64	2300	801	"	"
6	Turboatom	8.19	1030	2.46	1500	559	Steel	In operation
7	Turboatom	10.37	1100	2.73	1900	644	"	Project
8	Siemens	12.5	1146	3.03	2330	725	"	In operation
9	Siemens	16.0	1422	2.52	2160	786	Titanium alloy	Project
10	General Electric + Toshiba	11.68	1219	2.54	1880	678	Steel	In operation

Note: $\Omega = \pi d_m l$ is the last stage outlet area, l is the blade height, d_m/l is the parameter characterizing the blade height to mean diameter ratio, d_m is the mean diameter, d_r is the blade root diameter, and u_{per} is the blade circumferential speed at the periphery.

For tentatively estimating the power unit's electrical efficiency we assume that the boiler efficiency $\eta_b = 0.93$, the efficiency of power unit auxiliaries $\eta_{aux} = 0.95$, and the heat transport efficiency $\eta_{h.t} = 0.985$. In addition, in calculating the thermal circuit, the losses with the outlet velocity are taken equal to 0.015. With

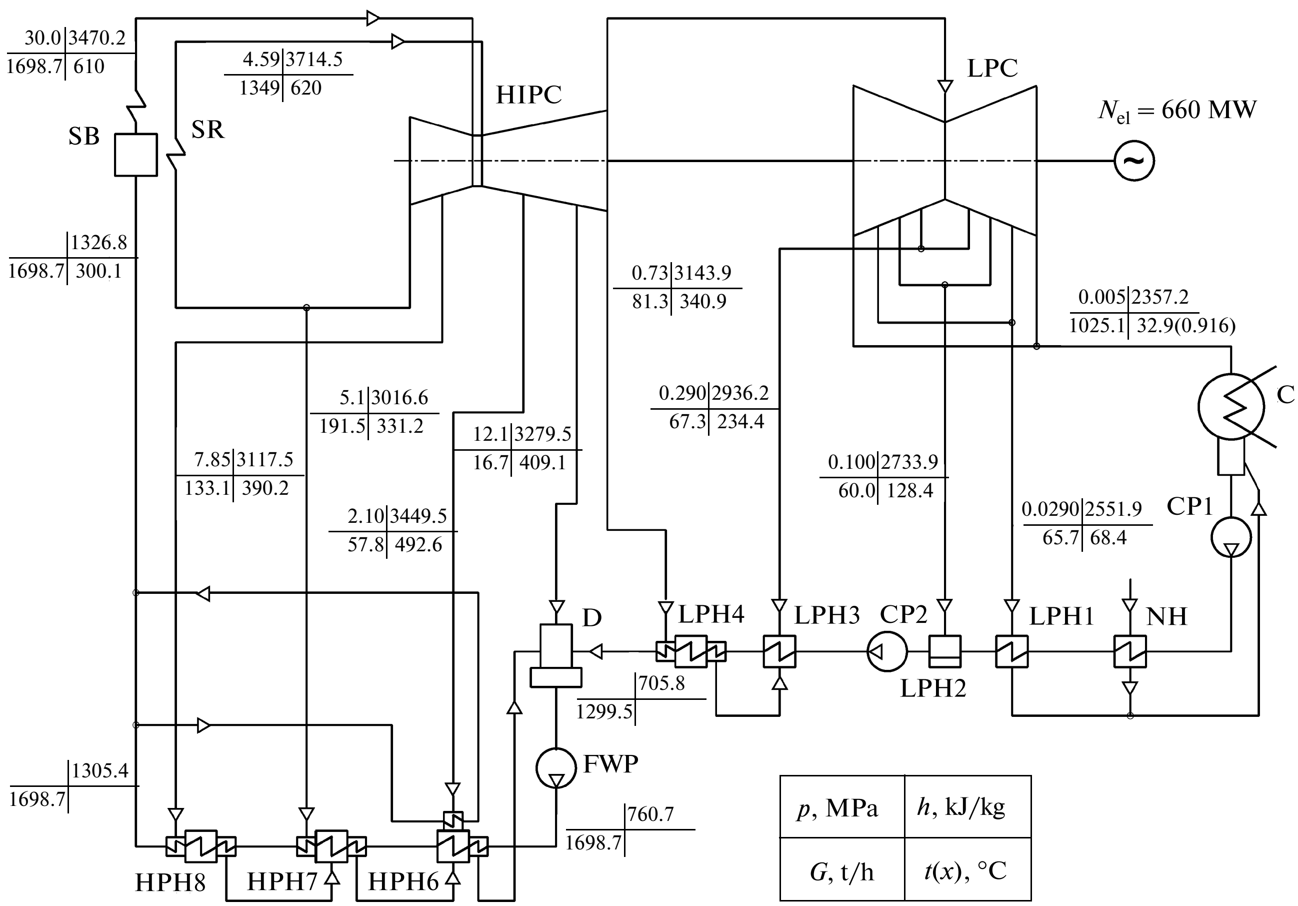

Fig. 2. Basic thermal circuit and main calculated parameters of the K-660-30 turbine unit. SB is the steam boiler, C is the condenser, CP1 and CP2 are the first and second lift condensate pumps, D is the deaerator, FWP is the feedwater pump, SR is steam reheating, LPH is the low-pressure heater, HPH is the high-pressure heater, and NH is the network heater.

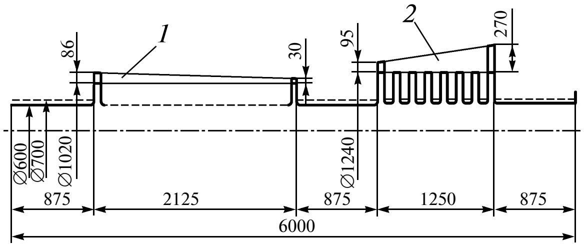

Fig. 3. Schematic design of the HIPC rotor. (*1*) and (*2*) are the HPP and IPP flow paths.

these assumptions, the electrical efficiency of the power unit equipped with the K-660-30 turbine for ultrasupercritical steam conditions will be

$$\eta_{p.u} = (3600/6852.1) \times 0.93 \times 0.95(1 - 0.015) \times 0.985 = 0.450.$$

Thus, the obtained efficiency value is in consistency with the design and operational data for power units that have been constructed.

How deviations of the main parameters (p_0, t_0, $t_{s.r}$, t_{fw}, and others) affect the heat consumption q_{gr} of the turbine unit is considered in [7]; however, it does not give any data on the influence of pressure in the condenser. Our calculations carried out for the K-660-30 turbine unit show that the power unit efficiency drops by around 1.37% as the pressure in the condenser increases from 3.5 to 5.0 kPa, i.e., at $p_c = 5$ kPa, $\eta_{p.u} = 0.444$.

Taking into account that many initial data and characteristics taken a priori contain some uncertainty, we consider it possible to say that the expected efficiency of the power unit for ultrasupercritical steam conditions equipped with a K-660-30 turbine will lie in the range $\eta_{p.u} = 0.45 \pm 0.015$.

The attainable efficiency of the K-660-30 turbine has been determined on the basis of estimated efficiencies of the flow paths used in the HPP, IPP, and LPP at the steam flowrates and parameters obtained from calculations of the thermal circuit carried out in accordance with the traditional procedure (from velocity triangles); the velocity coefficients φ and ψ were taken from the experimental data obtained at MEI and other organizations [8]. The estimations were carried out using all modern methods for improving efficiency, including meridian profiling in the HPP, saber-shaped blades in the IPP and LPP, use of advanced 3D calculations, modernized systems and types of peripheral and diaphragm seals, and modern methods of reducing losses due to moisture (in the LPP). The additional improvement of efficiency was taken into account on the basis of calculated and experimental data reported in the literature.

Attainable efficiency of the HPP and IPP. The following main parameters of the HIPC flow path were selected from preliminary calculations (Fig. 3):

number of stages in the HPP—10;

isoentropic difference of enthalpies in the HPP stages—50.3 kJ/kg (it was selected as the optimal one for the diameters indicated in Fig. 3);

axial sizes of the HPP rotor blades (that approximately equal to the chords of rotor blade profiles at the root) for the 1st to the 4th stage $b_{1-4} = 35$ mm, for the 5th to the 10th stage $b_{5-10} = 50$ mm (for such chords, the static bending stresses in the rotor blades of the 4th and 10th stages are equal to the adopted permissible value $\sigma_b = 35$ MPa; the stresses in the other stages are less than 35 MPa);

the R-2617A profile was chosen for the root sections of all rotor blades (from the atlas of profiles developed at MEI); the pitch of the rotor blades is close to its optimal value;

the axial distances between the HPP disks were selected by analogy with those existed in the turbines in operation [5];

the calculated value of the HPP middle stage's blade efficiency $\eta_{r.b}^{m} = 0.91$ [8];

the number of stages in the IPP is equal to 8;

the isoentropic difference of enthalpies in the IPP stages is equal to 75.75 kJ/kg (it was selected as the optimal one for the IPP dimensions shown in Fig. 3);

the axial sizes of the IPP rotor blades (that approximately equal to the rotor blade chords at the root) for stages from the 1st to the 3rd one $b_{1-3} = 55$ mm, for stages from the 4th to the 6th one $b_{4-6} = 60$ mm, and for the 7th and 8th stages $b_{7,8} = 65$ mm; the sizes of chords in each group are selected so that the static bending stresses in the last stage of each group (at the leading edge in the root section) were equal to their permissible values $\sigma_b = 35$ MPa (the stresses in the other stages are less than 35 MPa);

the axial distances between the IPP disks were selected by analogy with the stages of turbines that are in operation [5] and must be determined more exactly in carrying out a detailed calculation and design of the diaphragms.

The relative blade efficiencies of the IPP stages calculated from velocity triangles reach $\eta_{r.b} = 0.925$; the losses due to leaks are usually less than 1%. The use of saber blades and selection of blade profiles with reduced linear flowrates at the root and at the periphery make it possible to improve the IPP efficiency to $\eta_{r.b}^{IPP} \approx 0.94$, which is testified from the results of direct thermal tests carried out on IPPs under field conditions. For example, according to the data obtained from tests of the K-300-23.5 turbine No. 2 at the

Konakovo district power station, $\eta_{r.i} = 0.92–0.93$. It is exactly these efficiency values of the IPP flow part that were taken in calculating the thermal circuit.

Losses due to leaks in the HPP. Leaks of steam have an essential effect on the HPP efficiency. These leaks depend on the type of diaphragm and peripheral seals used in the turbine, and on other parameters, the ratio d_m/l_1 (where l_1 is the nozzle cascade height) and the equivalent gap δ_{eq} being the main ones of them. The generalizing dependences for calculating the leaks are given in [8].

An analysis of the available experience allows us to recommend the use of multicomb seals with variable pitches (VPMSs) as peripheral ones in the HPP for the considered turbine. These seals are distinguished by high degree of durability in operation (because large relative axial and radial displacements of the rotor and stator are allowed), small leaks due to using a large number of combs, and low level of nonconservative transverse aerodynamic forces provoking the occurrence of low-frequency steam vibration characteristic of high-pressure cylinders [9].

The peripheral seal of the HPP first stage is schematically shown in Fig. 4. Diaphragm seals are made in accordance with the traditional design of a staged multicomb seal.

Table 2 contains some initial data and results obtained from calculations of leaks for the HPP first, middle, and last stages. The calculations were carried out in accordance with the procedure described in [8], which is based on experimental studies of VPMSs carried out at the MEI Department of Steam and Gas Turbines [9].

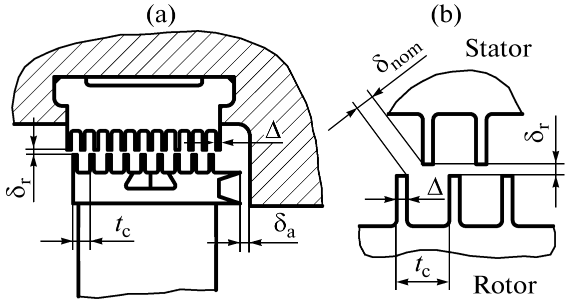

Fig. 4. Peripheral seal of the HPP first stage. δ_r, δ_a, and δ_{nom} are the radial, axial, and nominal gaps, and t_c and Δ are the comb pitch and thickness.

Proceeding from the data given in Table 2, we should recommend the use of VPMSs with nine combs on the rotor, twelve combs on the stator, radial gap equal to 1 mm, and diameter of diaphragm seals equal to 0.6 mm.

The losses due to leaks averaged over the entire high-pressure flow path are:

$$\text{for } d_d = 0.6 \text{ m}, \quad \zeta_l = \zeta_{l.p} + \zeta_{l.d} = 0.027, \text{ and}$$

$$\text{for } d_d = 0.7 \text{ m}, \zeta_l = 0.03.$$

Table 2. Losses due to leaks on the HPP of the K-660-30 turbine

Indicator	HPP stages		
	first	middle	last
Nozzle cascade height l_1, mm	28	56	84
Rotor cascade height, mm	30	58	86
Mean diameter, m	1.050	1.078	1.104
Number of combs on the rotor, pcs.	9	12	12
Number of combs on the stator, pcs.	12	15	15
Radial gap in the peripheral seal, mm	1.0	1.0	1.0
Nominal gap, mm	2.016	2.016	2.016
Equivalent gap of the peripheral seal, mm	0.356	0.309	0.309
Reactivity ratio	0.20	0.25	0.30
Loss due to leaks at the periphery $\zeta_{l.p}$	0.0323	0.0154	0.0118
Radial gap in the diaphragm seal, mm	0.74	0.74	0.74
Number of combs in the diaphragm seal, pcs.	16	14	10
Diaphragm seal diameter $d_d = 0.6$ m			
Equivalent gap of the diaphragm seal, mm	0.130	0.138	0.163
Loss due to leaks through the diaphragm seal $\zeta_{l.d}$	0.0110	0.0057	0.0044
Diaphragm seal diameter $d_d = 0.7$ m			
Loss due to leaks through the diaphragm seal $\zeta_{l.d}$	0.0148	0.0078	0.0060

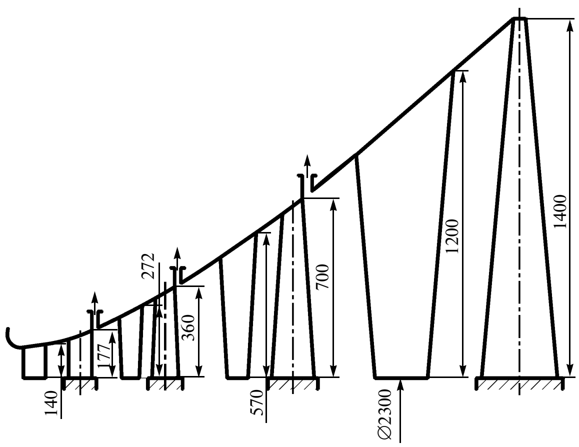

Fig. 5. Flow path of LPC of the K-660-30 turbine.

It can also be established that decreasing the radial gap from 1.0 to 0.5 mm (which is possible owing to the VPMS design) results in that the HPP efficiency increases by approximately 0.2% and by approximately 0.05% if the influence of two axial combs is taken into account (Fig. 4).

Expected efficiency of the high-pressure part. In estimating the HPP efficiency $\eta_{\mathrm{r.i}}^{\mathrm{HPP}}$ we assume that the blade efficiency $\eta_{\mathrm{r.b}}^{\mathrm{HPP}} = 0.91$, the heat return ratio $\alpha_{\mathrm{h}} = 0.01$, and that meridian profiling is used in the first five stages (in calculating the entire high-pressure flow path, this can give an increase in the efficiency by 1%). Then,

$$\eta_{\mathrm{r.i}}^{\mathrm{HPP}} = \eta_{\mathrm{r.b}}^{\mathrm{HPP}}(1 + \alpha_{\mathrm{h}})(1 - \zeta_{\mathrm{l}}) \times 1.01,$$

which for $d_{\mathrm{d}} = 0.6$ m gives $\eta_{\mathrm{r.i}}^{\mathrm{HPP}} = 0.91 \times 1.01(1 - 0.027) \times 1.01 = 0.90323$ and for $d_{\mathrm{d}} = 0.7$ m, $\eta_{\mathrm{r.b}}^{\mathrm{HPP}} = 0.91 \times 1.01(1 - 0.03) \times 1.01 = 0.90044$.

Hence, the shift from $d_{\mathrm{d}} = 0.6$ m to $d_{\mathrm{d}} = 0.7$ m results in that the efficiency drops by $(\Delta\eta/\eta)_{\mathrm{HPP}} \times 100 = (1 - 0.90044/0.90323) \times 100 = 0.3\%$.

Thus, the expected efficiency of the K-660-30 turbine's HPP is $\eta_{\mathrm{r.i}}^{\mathrm{HPP}} = 0.90$, which is by 1% lower than that taken in the calculation of the thermal circuit. It should be emphasized that the error of around 1% corresponds to the difference between the calculated value (obtained using the most advanced methods) and experimental data on the HPP efficiency.

Expected efficiency of the low-pressure part. Preliminary calculations were carried out, based on which the main parameters of the double-flow LPC's flow path shown in Fig. 5 have been selected. The LPC comprises four stages. The isoentropic difference of enthalpies in the LPP and some of its main parameters are given in Table 3.

The last stage of the LPC is made of advanced titanium rotor blades with the flow area $\Omega = 16.27$ m^2 (the blade is 1400-mm long, and the root diameter is equal to 2300 mm). Since the first two stages of the LPC operate in the region of superheated steam, and leaks in the LPC are less than 0.5%, it should be considered that the efficiency of these stages can be obtained at a level of 0.93.

The loss due to moisture is equal to around 1.20% in the last but one stage and 6.4% in the last stage. Therefore, the estimated efficiencies of these stages will be equal to around 0.91 and 0.84.

To reduce erosion of the last-stage rotor blades, means should be provided for internally heating the nozzle vanes of the last and possibly the last but one stages. The use of this measure makes it possible to completely eliminate coarsely dispersed moisture generated as liquid films are entrained from the nozzle vane surfaces. Heating of nozzle vanes seems to be the most efficient means for considerably reducing ero-

Table 3. Some characteristics of the LPP stages ($d_{\mathrm{r}} = 2300$ mm) of the K-660-30 turbine

Indicator	Stage number			
	1	2	3	4
Difference of enthalpies, kJ/kg	212	214	216	242
Reactivity ratio at the mean diameter	0.35	0.40	0.52	0.70
Rotor blade height, mm	177	360	700	1400
Steam quality at the stage outlet	1.0	1.0	0.978	0.916
Blade efficiency at the mean diameter estimated from stagnation parameters	0.93	0.94	–	0.91
Loss due to moisture (according to the MEI data), %	0	0	1.2	6.4
Loss due to moisture with the use of heating the last-stage nozzle vanes, %	0	0	1.2	3.4
Stage efficiency estimated from stagnation parameters:				
without taking into account the heating of nozzle vanes	0.93	0.94	–	0.84
with taking into account the heating of nozzle vanes	0.93	0.94	–	0.87

Table 4. Comparison between the indicators of the LMZ K-800-23.5-3 and K-660-30 turbine units

Turbine	Number of cylinders	Number of bearings in the turbine unit	Number of turbine stages	Turbine length, m
K-800-23.5-3	5	12	60	39.7
K-660-30	2	4	26	17.5

sion. In addition, a decrease in the amount of coarsely dispersed moisture results in smaller losses due to moisture. Assessments show that heating of the last-stage nozzle vanes makes it possible to reduce the losses due to moisture for the last stage by approximately 3%. It should be pointed out that Siemens Company actively uses this measure for reducing erosion of rotor blades.

Thus, by taking the effect of moisture and losses with outlet velocity into account, it is possible to determine more exactly the turbine unit's gross heat consumption at $p_c = 5$ kPa and the turbine unit efficiency η_t^{gr}: they are equal to 6999 kJ/(kW h) and 0.514. With such refinement and with the values of boiler efficiency, consumption for the power unit auxiliaries, and efficiency of heat transport taken above, we obtain that the power unit efficiency $\eta_{p.u} = 0.448 \approx 0.450$, which coincides with the value obtained earlier.

The results obtained from working out the predraft design of the K-660-30 turbine makes it possible to compare its main indicators with the indicators of the K-800-23.5-3 turbine of LMZ (Table 4).

The prospective turbine unit, the last-stage blade of which has the outlet area $\Omega = 16.27$ m^2, has an essential advantage over the traditional turbine unit. The use of the new turbine unit will make it possible to achieve a considerable reduction in the cost of the turbine and the power station as a whole.

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